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THESIS

THE EFFECT OF CIRCUMFERENTIAL TUBE WALL HEAT
CONDUCTION UPON LAMINAR FILMWISE CONDENSATION
ON THE OUTSIDE OF CONDENSER TUBES

by

Howard Michael Holland

December 1980

Thesis Advisor:

R. H. Nunn

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CONDUCTION UPON LAMINAR FILMWISE CONDENSATION
ON THE OUTSIDE OF CONDENSER TUBES

by

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Submitted in partial fulfillment of the
requirements for the degrees of

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ABSTRACT

This thesis describes the results of a theoretical study to predict the thermal behavior of an internally-cooled tube in a condensing vapor.

The analysis constitutes a unique application of the finite element method and provides new insights into the effects of circumferential conduction upon condenser tube performance. Comparisons are made between the present analysis and the theoretical and experimental works of others. The inclusion of circumferential conduction leads to an improvement in the predictive capabilities of the analytical model.

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I. INTRODUCTION

A. BACKGROUND INFORMATION

As the size, complexity, and hence the cost of modern warships have grown in the last thirty years, advances in basic research have been made and the computer has become a useful tool to the researcher and designer. The application of up-to-date research and computer technology to the design of modern warships is common but there remain major, costly components relatively unchanged and unreviewed in decades. It is in the interest of the Navy to ferret out these components and re-examine them.

One such component is the naval condenser. The basic engineering developments currently used to design naval condensers are from the Heat Exchanger Institute (H.E.I.) Standards for Steam Surface Condensers [1] and the standards of the Tubular Exchanger Manufacturers Association (T.E.M.A.) [2]. The Navy has further utilized the H.E.I. information in the Design Data Sheet (D.D.S.) [3]. While these standards have proven reliable, their performance margins and extensive use of averaged values may result in a larger, costlier condenser at no increase in performance and with an unnecessary margin of reliability. These performance margins, once necessary to insure reliability, are now called into question as the mysteries of condensation are unravelled.

It is for the purpose of reducing the size of the condenser that many researchers are trying to increase the heat transfer rate per unit of condensing surface area.

B. CONDENSATION IN A NAVAL CONDENSER

A typical naval condenser uses the heat transfer mechanism of filmwise condensation to the outside of horizontal, cylindrical tubes. Dropwise condensation is therefore not considered in this thesis.

The inside of the tube has single-phase turbulent forced convection heat transfer. The correlations associated with this phenomenon are accurate enough to justify confidence in their use. The fouling of the inside of the tube is poorly understood, and varies greatly, so that a necessary design allowance is made and a performance margin created. The heat transfer through the tube wall itself is by conduction, a classical problem in cylindrical coordinates, and can be calculated as accurately as the properties of the tube material and the surrounding heat transfer coefficients are known.

Aside from a greater knowledge of coolant-side fouling and how to control it, it is largely on the steam side of the tube that advances of knowledge can be reasonably expected. Here there are three important phenomena. The first is the transfer of heat through a thin layer of condensate from a vapor, either quiescent or flowing, which causes that vapor to lose its latent heat of vaporization and join the film of

condensate on the tube. As condensing particles join this liquid layer they retain some, none, or all of the momentum they possessed as a vapor. The dynamic interaction of the vapor and the liquid under both forced flow and quiescent vapor conditions presents a formidable problem to the researcher. A second phenomenon on the steam side of the tube is the presence of noncondensable gases which complicates and usually degrades the performance of the condenser. The third phenomenon is the inundation of a condenser tube by "condensate rain," or condensate condensed elsewhere in the condenser which, as it makes its way to the hotwell, impinges upon the condenser tube.

C. IMPROVING THE HEAT TRANSFER

If the rate of heat transfer for a given area of condensing surface were to be increased, then the total size of a condenser could be reduced. The result would be savings in space, weight, and possibly cost. Various methods of increasing the heat transfer in a naval condenser have been proposed. Among these are:

1. Reducing the Internal Fouling

The reduction of internal fouling would certainly increase the rate of heat transfer. While various tube metals and chemical treatments to the coolant side of the condenser can reduce fouling, operational and other considerations have prevented their use in naval condensers. Aside from mechanical cleaning, no acceptable method of significantly

reducing marine fouling in a naval condenser has presented itself.

2. Eliminating Noncondensable Gases

The elimination or reduction of noncondensable gases in the condensing steam would be expected to increase the heat transfer rate. It is a fact of realistic operation in a marine environment that noncondensable gases cannot be eliminated. There is room for improvement in their continual removal from the condenser by air ejector devices. As long as complicated systems with mechanical seals are used in the shipboard environment, some amount of noncondensable gases will be in the condenser.

3. Enhancing Heat Transfer

a. Shaping of Tubes

Various ingenious shapes of tubes have been demonstrated to increase heat transfer rate for a given amount of tube surface area. Complications arise from these tubes, however, not the least being a sharp increase in the pumping power required to maintain an adequate coolant flow rate.

b. Promoting Dropwise Condensation

The use of various promoters upon the tube outer surface to prevent the forming of a liquid film, thus keeping the condensate in droplet form, has been proposed and demonstrated by many authors. Since it is not the intention of this thesis to examine the effect of this phenomenon, suffice it to say that until now there remains a question as to whether

any promoter has demonstrated the robustness to withstand continued, long-term use.

4. Eliminating or Reducing Inundation

Since condensers are formed with banks of tubes, it is inevitable that condensate draining from one tube should impinge upon some other lower tube. The effect of this in filmwise condensation is normally to thicken the condensate film and to reduce the rate of heat transfer. The proper placement of collection trays or baffles is of great importance in channelling falling condensate away from lower tubes. The number and placement of these collection trays is not examined in this thesis.

5. Utilizing High Vapor Velocities

In many regions of a naval condenser, vapor velocities of 300 ft/sec or more may occur. These high velocities create a forced flow situation on the steam side, thin the film of condensate, and generally increase the heat transfer coefficients on the steam side. The higher velocities also cause complications, one of which is the separation of the vapor boundary layer causing the condensate film to thicken suddenly. The heat transfer in the region of the tube which is after the vapor phase separation point is relatively unknown.

D. REVIEW OF GENERAL RESEARCH IN CONDENSATION

In analyzing the condensation of steam on tubes, the cornerstone of research was laid by Nusselt [4] who analyzed

a quiescent vapor condensing on a vertical flat plate and on a horizontal cylinder. Nusselt's analysis allowed for no interfacial shear stress between the flowing condensate and the condensing vapor. He also used a constant tube wall surface temperature.

Rohsenow, et al., [5] later expanded this work to include the effect of vapor shear at the vapor-liquid interface upon the condensation rates. This analysis was for a vertical flat plate of uniform temperature with either turbulent or laminar flow of condensate.

Sparrow and Gregg [6] conducted the first analysis of surface condensation using boundary layer theory. They included momentum terms in the condensate film analysis to account for acceleration of the flowing condensate. They showed that the momentum terms were insignificant under most conditions.

Koh, et al., [7] did further work allowing the falling condensate to drag the vapor along with it, while still using a constant wall surface temperature.

Shekriladze and Gomelauro [8] expanded upon the earlier work of Rohsenow, et al., [5], analyzing the effect of a flowing vapor upon laminar filmwise condensation in the absence of significant gravity effects. Their model used a constant wall surface temperature.

Denney and Mills [9] expanded this area of analysis to include the influence of gravity upon the condensate film, holding to a constant wall surface temperature.

E. RECENT WORK CONCERNING CONDENSATION ON TUBES

1. Vapor-Condensate Interfacial Shear Models

One of the areas of continual interest in condensation of steam on a tube is the question of the effect of the shear or drag of the condensing vapor upon the flowing condensate. Theories which attempt to quantify the interaction abound, but recent ones take one of two general forms.

The first type of model says that total shear is the result of the summation of dry friction shear of the vapor upon a bare tube and a shear caused by the condensing vapor retaining some, none, or all of its vapor phase momentum. The deceleration of this vapor as it joins the liquid layer causes this shear. Various authors [8, 10, 11, 12, 13,] subscribe to some combination of these two shears.

Other researchers [14, 15] subscribe to a theory which reaches more deeply into the molecular level interactions and in reality appears to be a more minute examination of the same idea. Using a Reynolds Flux concept, these authors state that for a flowing fluid there is a transport of momentum in a direction normal to the mass flow which is on the molecular level in laminar flow but involves larger amounts of fluid in turbulent flow. This transport is a function of temperature in laminar flow and of temperature and free stream turbulence in turbulent flow. As one of these cross-flowing particles approaches a solid wall, it is decelerated to a standstill. As it rebounds from the wall, it is reaccelerated. Thus,

an overall rate of condensation can be represented by superimposing a velocity perpendicular to the wall, similar to suction through a dry wall.

The latter theory includes within it a relationship between the rate of condensation and temperature drop that must necessarily occur in the process [14]. The former frictional shear force theories commonly assume that the outer surface of the condensate film is at the steam saturation temperature.

2. Wall Temperature Models

Fujii, et al., [16] display from their experimental data that tube temperature varies around the outside surface of the tube. In their calculations, however, they use averaged values for tube wall temperature and for the temperature-dependent heat transfer coefficients to calculate heat transfer.¹

Nobbs and Mayhew [17] show that for steam flowing vertically downward, a variable temperature analysis results in a marked overall change in heat transfer. The state that earlier isothermal tube wall surface models were optimistic in their predicted heat transfer.

¹Since the writing of this thesis, new results have been obtained by Fujii and his coworkers. The reader is referred to the Proceedings of the Workshop on Modern Developments in Marine Condensers, Naval Postgraduate School, 26-28 March, 1980.

Nicol and Wallace [13] show that local heat transfer coefficients are indeed a function of local tube surface temperature and recognize the importance of the circumferential path in heat conduction. In one of the first analyses to include this path of heat transfer, these authors apply a relaxation technique and iterative scheme, changing the temperature distribution and then the local heat transfer coefficients. It is interesting to note that their results show that when the circumferential heat conduction path is included, the average heat transfer coefficient rises about twenty-five percent.

Nicol, et al., [18] analyze steam crossflow upon condenser tube (Fig. 1) using a formulation wherein local tube surface temperatures are used to calculate local heat transfer coefficients. Their results show that the variable wall temperature formulation predicts less heat transfer than older isothermal wall models. The effect of circumferential conduction around the tube wall is not included in their analysis. A close look at Fig. 7 of their work, partially reproduced here as Fig. 2, shows a theoretical wall temperature profile and a few measurements of surface temperature around the tube wall. In their discussion of this particular figure they state:

"The profiles are seen to agree quite well, the main differences being in the rounding off of the more extreme peaks of the theoretical profile because of the circumferential conduction in the tube wall, which is of course more pronounced in those regions where the temperature gradient is high."

Nicol, et al., [18] also present theoretical curves of local heat transfer coefficients, local heat flux, and local tube surface temperature. These cases are presented for both an older isothermal wall model and an anisothermal wall model where heat can flow only radially in the tube wall.

Fujii, et al., [19] use the results of Nicol, et al., [18] for variable wall surface temperature, coupled with earlier work [20] and develop another model based upon the assumption of uniform heat flux around the tube. These authors also recognize the path of circumferential heat conduction as an important area for further work.

F. Purpose of Research

It is in this final area that the author has analyzed, with the computer, a single tube of a condenser and has included in his analysis the circumferential conduction of heat in the tube wall, a phenomenon not customarily included in condenser heat transfer analysis.

II. A SPECIFIC PROBLEM FOR STUDY

A. THE SINGLE TUBE PROBLEM

A review of prior research has indicated ideas and trends. One basic idea was that local heat transfer coefficients depend upon local temperature differences and that the shearing force depends upon these heat transfer coefficients. The trend was away from older isothermal wall models to newer variable wall temperature (anisothermal) models. The logical continuation of the trend would indicate that a wall fully capable of two dimensional heat conduction would have been better still.

Since actual tube walls can conduct heat in the circumferential direction, then a model which could account for this would be an improvement over previous models. In order to test the plethora of shear force and condensation rate theories, a model including the effect of circumferential conduction is desirable, and may prove to be essential.

B. THE PROBLEM STATEMENT

The problem was posed for the upper half of a cylindrical condenser tube with a cross-flowing vapor, that is, where the direction of the gravitational force and the direction of vapor flow were perpendicular, (See Fig. 1a). Symmetry with the other half of the tube was assumed, allowing the simplification that the ends of the half tube, where it would normally

join the other half, were adiabatic walls. The problem for solution was set up so that θ , the local temperature difference, was defined thusly:

$$\theta(r, \phi) = T_s - T(r, \phi) \quad (1)$$

While T_s may vary with pressure changes around the tube, it is assumed constant herein. The governing equation for heat transfer within the tube wall is the Laplace Equation:

$$\frac{\partial^2 \theta}{\partial r^2} + \frac{1}{r} \frac{\partial \theta}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \theta}{\partial \phi^2} = 0 \quad , \quad (2)$$

with the following boundary conditions:

$$q_i = k_w \left(\frac{\partial \theta}{\partial r} \right)_{r_i, \phi} = h_i [\theta(r_i, \phi) - \theta_c]$$

$$q_o = -k_w \left(\frac{\partial \theta}{\partial r} \right)_{r_o, \phi} = h_l \theta_o .$$

On the inside of the tube turbulent forced convection was assumed and the Dittus-Boelter correlation [21] was used to provide a heat transfer coefficient.

On the outside of the tube, the condensate flow was taken to be laminar, and heat transfer through the thin condensate layer was assumed to be purely by conduction. For such conditions the film heat transfer coefficient is given by:

$$h_l = \frac{k_l}{\delta} \quad (3)$$

The film thickness δ was calculated from the formulation due to Nicol, et al., [18] (see also Appendix A). This analysis yields the following first-order differential equation:

$$\frac{d\delta}{d\phi} = \frac{\frac{3\mu r_o k_l \theta_o}{gh_{fg} \rho_l (\rho_l - \rho_v)} - \delta^4 \sin \phi - \frac{3\delta^3}{2g(\rho_l - \rho_v)} \frac{d\tau_v}{d\phi}}{\frac{3\delta^2 \tau_v}{g(\rho_l - \rho_v)} - 3\delta^3 \cos \phi} \quad (4)$$

The shear term, τ_v , used by Nicol, et al., [18] was one given by Schlichting [22] and is of the following form:

$$\tau_v = \frac{\frac{1}{2}\rho_v U_v^2}{\left(\frac{U_v r_o \rho_v}{\mu_v}\right)^{1/2}} (6.973\phi - 2.732\phi^3 + 0.292\phi^5 - 0.0183\phi^7 + 0.000043\phi^9 - 0.000115\phi^{11}). \quad (5)$$

For the sake of comparison this author used the same shear stress model. This model calculates vapor boundary layer separation at about 108 degrees from the forward stagnation point. Thereafter, the shear stress was set to zero.

In order to integrate Eq. (4) a starting value of condensate film thickness at the stagnation point was needed. Because high velocity steam flowing horizontally normal to a horizontal tube was to be studied, $\frac{d\delta}{d\phi}$ was set equal to zero for the stagnation point ($\phi = 0$), and an approximate starting value for the condensate film thickness, δ , was found from Eqs. (4) and (5).

Once this starting value was in hand, Eq. (4) was integrated, in its entirety, around the tube using an estimate of the tube surface temperature difference, $[\theta_o = \theta(r_o, \phi)]$. The integration resulted in the local film thickness around the tube wall. Local heat transfer coefficients for the steam side were then calculated according to Eq. (3).

Using these local heat transfer coefficients Eq. (2) could then be integrated, resulting in a temperature distribution throughout the solid tube wall. With a new set of tube-wall outer surface temperatures in hand, a new set of local heat

transfer coefficients could be calculated and applied for another iteration of the Laplace equation. If temperatures converged and heat transfer balanced, then a solution was in hand.

C. THE GALERKIN FINITE ELEMENT METHOD

The purpose of this application of the Galerkin Finite Element method is to solve a steady state two dimensional heat conduction problem by converting the governing partial differential equation, the Laplace equations in this case, to a series of algebraic equations. Each of these algebraic equations represents the partial differential equation over a small sub-region of the total domain called an element. The result of the simultaneous solution of these algebraic equations is to drive the integral square error of the approximate solution to zero over each element. Elements may be of different sizes and may be clustered in areas where greater accuracy is desired.

The Galerkin Finite Element method differs from the Galerkin method in that, since the algebraic equation representing the approximate solution exists over only one small element, the resultant matrix of algebraic equations for solution takes on a banded form. That is, it has zero upper and lower triangles of significant size. The resultant savings in computer storage space and computational time by using special matrix equation solvers is significant.

In this discreet approximation method the boundary conditions are applied locally to each element on the boundary of the domain. In this case linear triangular elements were used. The algebraic approximations to the solution, called basis or shape functions, were linear. The resultant approximate solution amounts to approximating a smooth three dimensional surface by a mosaic containing a large number of small triangles. Triangles size varied, small ones being used in such areas of interest as the forward stagnation point and anywhere a steep temperature gradient could be expected to exist. Larger elements were used in areas where the temperature profile was considered to be relatively unchanging. As solutions progressed, refinement of the triangular meshes highlighted and concentrated upon areas of interest.

For greater detail on both the Galerkin method and the Galerkin finite element method, the interested reader is referred to Ozisik [23] and Fairweather [24].

The formulation and computer program were tested using classical two-dimensional problems of heat conduction from Carslaw and Jaeger [25]. The first of these was a rectangle with one side at a given temperature and the other three at a specified lower temperature. The second was a rectangle with two adjoining sides adiabatic, a third side at a given temperature, and the fourth having a given heat transfer

coefficient to a medium at some lower temperature. In both cases the finite element solutions converged to exact solutions.

A further test of the program and formulation was the problem of a rectangle with two opposite sides adiabatic, a third side at a given temperature, and the fourth side having a local heat transfer coefficient to some medium at a lower temperature. This local heat transfer coefficient was dependent upon both the local surface temperature and the position along the wall. Since this problem cannot be compared to a classical solution, any solution could only be given credence by detailed examination. It was seen that the solution converged for smaller and smaller triangle sizes, that plots of isotherms internal to the rectangle were smooth and not obviously in error, and that heat transfer into and out of the rectangle was in balance. While certainly no proof, these comparisons led to confidence in the program.

D. APPLICATION OF THE INNER BOUNDARY CONDITION

For the inside of the tube, the boundary condition of convection heat transfer, Eq. (2), can be simply represented by noting that the inside film coefficient, h_i , is given by the Dittus-Boelter correlation [21], and is a function of coolant bulk temperature and flow rate. Thus, since the thermal resistance at this interface is independent of local tube wall temperature, the resistance may be represented by a fictitious

thickness of insulating material according to the relationship:

$$R = \frac{1}{h_i A} = \frac{r_i \ln \left(\frac{r_i}{r_i - t_f} \right)}{K_f A} . \quad (6)$$

From this, the thickness of the fictitious ring t_f , was determined and the equivalent boundary condition, that the coolant temperature existed at the inside of the fictitious ring, was imposed.

E. SOLUTION PROCEDURE

On the outside of the tube, the local heat transfer coefficients were computed all around the tube for the current value of the calculated local outside wall temperature. These local heat transfer coefficients were applied to elements having sides on the outer tube wall.

In order to converge to the proper wall temperatures and temperature-dependent heat transfer coefficients, an iterative scheme was followed wherein a new set of local heat transfer coefficients for the outside of the tube were computed from new solution temperatures. The process was repeated until temperatures at all nodal points changed less than 0.02 degrees Celsius for consecutive iterations.

Having stabilized the local temperatures throughout the tube wall, a check was made to insure that the heat transfer balanced. The total heat deposited upon the condenser tube outer wall by the condensing steam was compared to the total heat conducted into the tube wall at the outer surface and to the total heat conducted from the tube at its inner surface.

F. COMPARISON WITH PREVIOUS SOLUTIONS

The aforementioned paper of Nicol, et al., [18] provided fertile ground for comparison, but only if the test conditions could be simulated. These authors were able to achieve a given average tube outer surface temperature by varying the flowrate of the coolant and hence the internal heat transfer coefficient. In the finite element formulation this condition is obtained by varying the thickness of the aforementioned fictitious ring. In order to preclude significant circumferential conduction in the fictitious ring, the circumferential thermal conductivity of the fictitious ring was chosen to be one one-hundredth of the thermal conductivity of the tube wall.

III. RESULTS

A. TUBE WALL TEMPERATURE

The tube wall temperature predictions of the author's finite element model are shown in Fig. (3), and are seen to support the prediction of Nicol et al., [18] that circumferential heat conduction would substantially round off the theoretical temperature profile obtained by considering radial heat conduction only. It is also worthwhile to note that the maximum and minimum temperatures are less extreme than those predicted by the radial heat flow model of Nicol et al., [18] and generally agree more closely with their experimental results. While the actual differences in temperature appear small, near the separation point of the vapor flow these differences approach thirty percent.

Nicol et al., [18] also show a theoretical tube wall surface temperature profile for another average wall temperature. This profile is reproduced in Fig. 4 along with a theoretical profile developed from the author's model. As before, the steep gradients are rounded off and the overall temperature profile is less extreme.

A plot of the isotherms within the tube wall is quite revealing and is shown in Fig. (5). The direction of vapor flow and point of vapor boundary layer separation, as indicated by the thickening of the condensate layer, are shown on the

figure. The isotherms are seen to start close together near the vapor stagnation point and to spread, reflecting lower rates of heat transfer, progressively around the tube. In the vicinity of separation of the vapor boundary layer, the isotherms are seen to deviate significantly from their concentric upstream orientation. This reflects a significant amount of heat transfer circumferentially around the tube in this general vicinity. The much wider spacing of the isotherms on the back part of the tube indicates that according to this model it is an area of relatively low heat transfer rate.

1. Local Heat Transfer Coefficients

Figure 6 shows a comparison of local heat transfer coefficients around the tube wall. The isothermal wall and radial heat flow local transfer coefficients of Nicol et al., are partially reproduced. The results of the finite element model are seen to lie generally between the other two models with the exception of the vapor-separated region. In the region prior to separation, as seen in Fig. 4, the wall temperature difference of the finite element model lies between the isothermal wall case and radial heat flow case. From Eq. (4) at the stagnation point, the initial value of the film thickness is proportional to $\theta_0^{1/3}$. Therefore, a larger temperature difference will create a larger initial film thickness, and therefore a smaller heat transfer coefficient, as given by Eq. (3). It is not unexpected that this formulation and solution should predict a greater heat

transfer coefficient in the separated region than the isothermal wall model. Since there was a higher wall temperature and therefore less steam condensed in the unseparated portion of the tube in the present model, there is a thinner condensate layer in the after section of the tube. In the separated region the greater heat transfer coefficient predicted by the author's model over the radial heat flow model is not fully explained, but is no-doubt due, in part, to the assumption of symmetry in the present formulation.

2. Local Heat Flux

In still a third examination under these same conditions, Nicol et al. in their Fig. 6 present local heat flux for both the isothermal wall and radial heat flow models. These profiles are partially reproduced here in Fig. 7 along with the author's results which again, as expected, lie between the two previous models except in the separated-vapor region of the tube. It is interesting to note that the inclusion of circumferential effects generates a heat flux which is much closer to being uniform than that which is obtained with the isothermal wall model.

IV. CONCLUSIONS

The circumferential path of heat conduction has been successfully included in the solution of laminar filmwise condensation of high velocity steam on condenser tubes. The author's anisothermal fully-conducting wall model yields closer agreement to previously measured tube wall surface temperatures than a previous anisothermal wall model which allowed only radial heat flow. The maximum and minimum values of wall temperature were less extreme for the anisothermal fully conducting wall than for the anisothermal wall with only radial heat flow.

The addition of a circumferential path for heat flow results in a complicated interplay between this heat flow and heat being deposited locally by condensation and leaving in the internal cooling water. The addition of vapor velocity effects on the outside surface of the film, and especially the sudden changes arising from separation of the vapor flow result in a highly complex sequence of thermal and fluid-dynamical interactions. The methods developed herein, though still subject to comprehensive experimental verification, are adequate to predict these complex phenomena.

V. RECOMMENDATIONS

Now that a model for two-dimensional heat conduction which includes temperature-dependent local heat transfer coefficients has been developed and demonstrated, the use of this model for further study is suggested. Areas thought fruitful for further examination are both the shear stress models and the temperature-dependent condensation rate models. For shear stress models the separated vapor region or trailing area must be given closer examination since it accounts for about forty percent of the surface area of the tube and the fluid flow characteristics in this region remain relatively unknown. In the study of temperature-dependent condensation rate models, further examination of the effect of the velocity of the flowing vapor and local temperature is expected to be fruitful.

The vigorous exercise of this model through a broad range of steam-side and coolant side conditions is suggested.

The model of condensation with constant heat flux proposed by several authors [18, 20, 26], should be re-examined since circumferential heat conduction in the tube wall, when included in the analysis, seems to yield a nearly constant heat flux distribution.

APPENDIX A

Local Rate of Change of Condensate Film Thickness

A force balance on the shaded element of Fig. 1b yields the following:

$$\tau_v r_0 d\phi - \tau_\ell r_0 d\phi - g(\rho_\ell - \rho_v) \cos\phi r_0 (\delta - y) d\phi = 0$$

so that:

$$\tau_\ell = \mu \frac{\partial u}{\partial y} = \tau_v - (\rho_\ell - \rho_v) g \cos\phi (\delta - y) .$$

Integrating:

$$u = \frac{\tau_v}{\mu} y - (\rho_\ell - \rho_v) \frac{g}{\mu} \cos\phi \left(\delta y - \frac{y^2}{2} \right) .$$

The average velocity, \bar{u} is:

$$\begin{aligned} \bar{u} &= \frac{1}{\delta} \int_0^\delta u \, dy = \frac{1}{\delta} \frac{\tau_v}{\mu} \frac{\delta^2}{2} - \frac{g}{\mu} \left(\frac{\rho_\ell - \rho_v}{\delta} \right) \cos\phi \left[\frac{\delta^3}{2} - \frac{\delta^3}{6} \right] , \text{ or} \\ \bar{u} &= \frac{\tau_v \delta}{2\mu} - (\rho_\ell - \rho_v) \frac{g}{\mu} \cos\phi \frac{\delta^2}{3} . \end{aligned} \quad (8)$$

The mass flow rate of condensate around the tube is:

$$\dot{m}_\ell = \rho_\ell \delta \bar{u}$$

and, with Eq. (8),

$$\dot{m}_\ell = \frac{\tau_v \rho_\ell}{2\mu} \delta^2 - \frac{g \rho_\ell (\rho_\ell - \rho_v)}{3\mu} \cos\phi \delta^3 .$$

The rate of change of mass flow rate with respect to angle is:

$$\begin{aligned} \frac{d\dot{m}_\ell}{d\phi} &= \frac{\rho_\ell \delta^2}{2\mu} \frac{d\tau_v}{d\phi} + \frac{\rho_\ell \tau_v \delta}{\mu} \frac{d\delta}{d\phi} + \frac{g \rho_\ell (\rho_\ell - \rho_v) \delta^3 \sin\phi}{3\mu} \\ &\quad - \frac{g \rho_\ell (\rho_\ell - \rho_v)}{\mu} \cos\phi \delta^2 \frac{d\delta}{d\phi} \end{aligned}$$

Since $d\dot{m}/d\phi$ can also be equated to the amount of steam condensing, the following relationship applies:

$$\frac{d\dot{m}_\ell}{d\phi} = \frac{r_o k_\ell \theta_o}{\delta h'_{fg}}$$

where h'_{fg} is the latent heat of vaporization corrected for subcooling. By making this substitution and solving for $d\delta/d\phi$, the relationship given in Eq. (4) is obtained.


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A FINITE ELEMENT PROGRAM USING FIRST ORDER LINEAR GALERKIN
FINITE ELEMENTS TO SOLVE TWO-DIMENSIONAL HEAT TRANSFER
PROBLEMS WITH NON-LINEAR, TEMPERATURE DEPENDENT BOUNDARY
CONDITIONS.

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      THE PRESENT SET UP IS FOR A CYLINDRICAL CONDENSER TUBE
      WITH FILMWISE CONDENSATION ON THE OUTSIDE AND TURBULENT
      FORCED CONVECTION ON THE INSIDE.

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      THE FOLLOWING IS A LIST OF
      VARIABLE NAMES USED IN THE
      PROGRAM.

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      VARIABLES OF THE GENERAL PROBLEM.

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      ATRI      THE AREA OF EACH TRIANGLE IN THE FINITE ELEMENT
      CSUEP      VALUE OF SPECIFIC HEAT OF LIQUID WATER AT CONSTANT
      FNEEC      PRESSURE.
      F          VALUE OF THE FORCING FUNCTION AT EACH ESSENTIAL
      BOUNDARY CONDITION.
      F          A VECTOR OF RIGHT HAND SIDES FOR SOLVING THE MATRIX
      EQUATION RESULTING FROM THE FINITE ELEMENT
      FORMULATION.
      LCCAL      LOCAL HEAT TRANSFER COEFFICIENT.
      KCCRR      LOCAL HEAT TRANSFER COEFFICIENT.
      NAX        THE CORRESPONDENCE TABLE RELATING THE POSITIONS
      OF THE THREE CORNERS OF EACH TRIANGLE.
      NCOLN      THE TOTAL BANDWIDTH OF THE BANDED GLOBAL MATRIX,
      A COUNTER.

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VAT00001C
VAT00002C
VAT00003C
VAT00004C
VAT00005C
VAT00006C
VAT00007C
VAT00008C
VAT00009C
VAT00010C
VAT00011C
VAT00012C
VAT00013C
VAT00014C
VAT00015C
VAT00016C
VAT00017C
VAT00018C
VAT00019C
VAT00020C
VAT00021C
VAT00022C
VAT00023C
VAT00024C
VAT00025C
VAT00026C
VAT00027C
VAT00028C
VAT00029C
VAT00030C
VAT00031C
VAT00032C
VAT00033C
VAT00034C
VAT00035C
VAT00036C
VAT00037C
VAT00038C
VAT00039C
VAT00040C
VAT00041C
VAT00042C
VAT00043C
VAT00044C
VAT00045C
VAT00046C
VAT00047C
VAT00048C

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VAT00490
VAT00500
VAT00510
VAT00520
VAT00530
VAT00540
VAT00550
VAT00560
VAT00570
VAT00580
VAT00590
VAT00600
VAT00610
VAT00620
VAT00630
VAT00640
VAT00650
VAT00660
VAT00670
VAT00680
VAT00690
VAT00700
VAT00710
VAT00720
VAT00730
VAT00740
VAT00750
VAT00760
VAT00770
VAT00780
VAT00790
VAT00800
VAT00810
VAT00820
VAT00830
VAT00840
VAT00850
VAT00860
VAT00870
VAT00880
VAT00890
VAT00900
VAT00910
VAT00920
VAT00930
VAT00940
VAT00950
VAT00960

NCELI A VECTOR CONTAINING THE NUMBER OF EACH CONVECTION
NCELG ELEMENT ON THE INSIDE OF THE DOMAIN.
NCHK A VECTOR CONTAINING THE NUMBER OF EACH CONVECTION
NCELG ELEMENT ON THE OUTSIDE OF THE DOMAIN.
NEBC A VALUE USED TO CHECK AGREEMENT FROM ITERATION TO
ITERATION.
NEL THE NUMBER OF EACH OF THE NODAL POINTS WHERE A
NLC TEMPERATURE IS SPECIFIED.
NNP THE TOTAL NUMBER OF ELEMENTS.
NSDC THE NUMBER OF LOWER CO-DIAGONALS OF THE GLOBAL MATRIX
IN Banded STORAGE FORM.
NUC THE TOTAL NUMBER OF NODAL POINTS.
RHOH FOR EVERY ELEMENT HAVING A CONVECTION BOUNDARY, THIS
X IS THE NUMBER OF SIDES ON THE BOUNDARY.
RHOL THE NUMBER OF UPPER CO-DIAGONALS OF THE GLOBAL MATRIX
IN Banded STORAGE FORM.
RHOV THE DENSITY OF LIQUID WATER AT SATURATION CONDITIONS.
X THE DENSITY OF WATER VAPOR AT SATURATION CONDITIONS.
Y THE CRITICAL COORDINATES IN THE CARTESIAN COORDINATE
SYSTEM.
XLO12 THE VERTICAL COORDINATE IN THE CARTESIAN COORDINATE
SYSTEM.
XLO23 THE LENGTH OF THE SIDE OF A CONVECTION ELEMENT FROM
LOCAL NODAL POINT 1 TO LOCAL NODAL POINT 2.
XL THE LENGTH OF A CONVECTION ELEMENT
TBEAR FROM LOCAL NODAL POINT 2 TO LOCAL NODAL POINT 3.
Z A CHECKSPACE FOR THE LINEAR EQUATION SOLVER.
ZAST THE AVERAGE OF LOCAL TEMPERATURE ALONG THE CONVECTIVE
BOUNDARY OF AN ELEMENT HAVING A CONVECTIVE
BOUNDARY.
XKLIQ THE GLOBAL MATRIX FORMED IN THE FINITE ELEMENT METHOD,
STORED IN Banded STORAGE FORM.
ZAST A STORAGE PLACE FOR THE GLOBAL MATRIX SO THAT ONCE IT IS
INITIALLY FORMED IT NEEDS NOT BE FORMED AGAIN FOR
EACH SUCCESSIVE ITERATION.
XKLIQ THE THERMAL CONDUCTIVITY OF LIQUID WATER AT SATURATION
CONDITIONS.
XKSOL THE THERMAL CONDUCTIVITY OF THE SOLID TUBE MATERIAL.
XMUL THE DYNAMIC VISCOSITY OF LIQUID WATER AT THE SATURATION
CONDITIONS.
XMUV THE DYNAMIC VISCOSITY OF WATER VAPOR AT THE SATURATION
CONDITIONS.
XHFG THE LATENT HEAT OF VAPORIZATION OF WATER AT THE
SATURATION CONDITIONS.
VELST THE FREE STREAM VELOCITY OF THE FLOWING STEAM.

CC

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VARIABLE NAMES ASSOCIATED WITH
THE SPECIFIC GEOMETRY OF THIS
PROBLEM.

R THE VARIOUS RADII AT WHICH POINTS WILL BE PLACED IN
THE FINITE ELEMENT GRID.
PTEE THE VARIOUS ANGLES AT WHICH PCINTS WILL BE PLACED IN
THE FINITE ELEMENT GRID.
TAU A VECTOR OF LOCAL SHEAR FORCE VALUES, ONE FOR EACH CF
THE CONVECTION ELEMENTS.
DTAL A VECTOR OF LOCAL RATE OF CHANGE OF SHEAR FORCE WITH
RESPECT TO ANGLE PTEE FOR EACH CONVECTION ELEMENT.
NBNPI NUMBER OF BOUNDARY NODAL POINTS, INSIDE.
NBNPFC NUMBER OF BOUNDARY NODAL POINTS, OUTSIDE.
NANG NUMBER OF ANGLES AT WHICH NODAL PCINTS ARE TO BE
PLACED.
NRAD NUMBER OF RADIALLY ORIENTED HEAT FLOW PATHS.
NEIR THE NUMBER OF ELEMENTS IN A RADIAL ROW.
NPIR THE NUMBER OF NODAL PCINTS IN A RADIAL ROW.
NCBCG THE NUMBER OF OUTER CONVECTION BOUNDARY CONDITIONS.

DIMENSION F(320), NCELC(40), NCELI(40), NSDC(40), HCK(600)
1, TEM(320), TEAR(40), YEAR(40), FH(600)
DIMENSION KCR(600, 3), ZA(320, 25), R(15), FTEE(50)
DIMENSION XL(8000), ZAST(320, 25), TAU(40), DTAL(40)
REAL * E X(320), Y(320), ATRI(600), XLO12(40), XLO23(40)
DIMENSION C(10), NEBC(40), FNEBC(40), NBNPI(40), NBNPC(40)
READ(5, 11) NLC, NUC, NCCUN, NCHK, NCD
READ(5, 11) NNP, NEL, CULBF, FHCL, RHOV, XKLIC, XKSCL
READ(5, 11) XMLV, XHFG, VELST
FORMAT(12, F16.14), F16.2, F10.5)
FORMAT(5, I2) NBC
FORMAT(15)
DO 146 IE=1, NBC
READ(5, 142) NEBC(IE), FNEEC(IE)
FORMAT(15, F10.5)
CCNTIN(E

106

120

142

146

CVERT IS A SUBROUTINE WHICH READS IN VALUES OF ANGLE
FROM THE FORWARD STAGNATION POINT AND RADII FROM THE
CENTER OF THE TUBE AND CONVERTS THESE TO X AND Y CARTESIAN
COORDINATES.

CCCCCCCCCCCC

VAT00970
VATCC980
VAT00990
VAT01000
VATC1010
VAT01020
VAT01030
VAT01040
VAT01050
VATC1060
VAT01070
VAT01080
VATC1090
VATC1100
VAT01110
VAT01120
VAT01130
VAT01140
VAT01150
VATC1160
VAT01170
VAT01180
VATC1190
VATC1200
VAT01210
VATC1220
VAT01230
VAT01240
VATC1250
VAT01260
VAT01270
VAT01280
VAT01290
VATC1300
VAT01310
VATC1320
VATC1330
VATC1340
VAT01350
VAT01360
VAT01370
VAT01380
VAT01390
VAT01400
VATC1410
VAT01420
VATC1430
VATC1440

CC CC

CALL CVERT(X,Y,R,PTEE,NNF)

KCRRF IS A SUBROUTINE USED TO DEVELOP THE CORRESPONDENCE
TABLE IN ORDER TO CONNECT THE NODAL POINTS IN THE PROPER ORDER
TO FORM THE PROPER TRIANGULAR GRID.

CALL KCRRF(KCRR,NEL)

GECM IS A SUBROUTINE USED TO FIGURE THE AREA OF EACH
TRIANGULAR ELEMENT AND THE LENGTH OF ANY SIDE ALONG A CON-
VECTION BOUNDARY CONDITION.

CALL GECM(NNF,NEL,X,Y,KCRR,ATRI,NCELO,XLO12,XLO23,
INCC,NSCC)

FCRMB IS A SUBROUTINE WHICH TAKES THE X AND Y COORDINATES
AND THE AREA OF EACH TRIANGLE AND CONSTRUCTS THE LOCAL THREE BY
THREE MATRIX FOR EACH ELEMENT. IT THEN SUPERIMPOSES EACH OF
THE VALUES FROM THE LOCAL THREE BY THREE MATRIX UPON THE PROPERLY
NUMBERED NODAL POINTS ACCORDING TO THE CORRESPONDENCE TABLE.

CALL FCFME (ATRI,NEL,NNP,X,Y,ZA,KORR,NLC,NLC)

ZASTC IS A SUBROUTINE WHICH STORES THE GLOBAL MATRIX, ZA,
SO THAT IT NEED NOT BE REFORMED FOR EACH SUCCESSIVE ITERATION.

CALL ZASTC(NNP,ZA,ZAST,NLC,NLC)

VAT0145C
VAT0146C
VAT0147C
VAT0148C
VAT0149C
VAT0150C
VAT0151C
VAT0152C
VAT0153C
VAT0154C
VAT0155C
VAT0156C
VAT0157C
VAT0158C
VAT0159C
VAT0160C
VAT0161C
VAT0162C
VAT0163C
VAT0164C
VAT0165C
VAT0166C
VAT0167C
VAT0168C
VAT0169C
VAT0170C
VAT0171C
VAT0172C
VAT0173C
VAT0174C
VAT0175C
VAT0176C
VAT0177C
VAT0178C
VAT0179C
VAT0180C
VAT0181C
VAT0182C
VAT0183C
VAT0184C
VAT0185C
VAT0186C
VAT0187C
VAT0188C
VAT0189C
VAT0190C
VAT0191C
VAT0192C


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TEMDS IS A SUBROUTINE WHICH DISTRIBUTES INITIAL VALUES OF
TEMPERATURE DIFFERENCE (FIRST GUESSES) AT THE CONVECTION
ASSOCIATED NODAL POINTS SO THAT THE LOCAL TEMPERATURE DEPENDENT
HEAT TRANSFER COEFFICIENTS CAN BE CALCULATED THE FIRST TIME.
VATC193C
VATC194C
VATC195C
VATC196C
VATC197C
VATC198C
VATC199C
VATC200C
VATC201C
VATC202C
VATC203C
VATC204C
VATC205C
VATC206C
VATC207C
VATC208C
VATC209C
VATC210C
VATC211C
VATC212C
VATC213C
VATC214C
VATC215C
VATC216C
VATC217C
VATC218C
VATC219C
VATC220C
VATC221C
VATC222C
VATC223C
VATC224C
VATC225C
VATC226C
VATC227C
VATC228C
VATC229C
VATC230C
VATC231C
VATC232C
VATC233C
VATC234C
VATC235C
VATC236C
VATC237C
VATC238C
VATC239C
VATC240C

CALL TEMDS(TEM,NBNPC,NCANPD)

FLCCC IS A SUBROUTINE WHICH FIGURES THE LOCAL HEAT TRANSFER
COEFFICIENTS ON THE OUTSIDE OF THE TUBE. THIS SUBROUTINE IS
INCLUDED IN ALL ITERATIONS.

CALL FLCCC(XHFG,RHCL,RHCV,XMUL,XMLIG,XKSCL,PHEE,R,TEM,HOK,
INCELC,KCRR,NCG,TBAR,FF,FEEDBAR,VELST,CSLBP)
TFAC1=C.
AR=C.
DO 324 JL=1,NCG
TFAC1=TFAC1+TBAR(JU)*XLC12(JU)
AR=AR+XLC12(JU)
CONTINUE
TAV=TFAC1/AR
WRITE(6,165)TAV
WRITE(7,165)TAV
FORMAT(' AVERAGE TEMF IS',F10.5)

FORMF IS A SUBROUTINE WHICH PLACES ZEROS IN THE RIGHT HAND
SIDE OF THE MATRIX EQUATION.

CALL FORMF(F,NNP)

FORMCD IS A SUBROUTINE WHICH MODIFIES THE GLOBAL MATRIX FOR
THE CONVECTION BOUNDARY CONDITIONS.

CALL FORMCD (NNP,ZA,KCRR,NCELC,NSDO,XLC12,XLC23,FCR,NC0
1,NLC,NLC)

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CCCCCCCC CCCCCCCCC CCCCCCCCC CCCCCCCCC

MCBCE IS A SUBROUTINE WHICH MODIFIES THE GLOBAL MATRIX AND
THE FORCING FUNCTION FOR THE SPECIFIED TEMPERATURES (ESSENTIAL
BOUNDARY CONDITIONS.)

CALL MCBCE(ZA,F,NNP,NCCUN,NEBC,NBC,FNEBC,NLC,NUC)
NAX=NLC+1

LECTIE IS A SINGLE PRECISION LINEAR EQUATION SOLVER
SPECIFICALLY PROGRAMMED TO WORK UPON BANDED MATRICES.

CALL LECTIE(ZA,NNP,NLC,NUC,320,F,1,1,0,XL,IER)

CLOSE IS A SUBROUTINE WHICH CHECKS AFTER EACH ITERATION TO
SEE HOW CLOSE THE NEW TEMPERATURE DIFFERENCES AGREE WITH THE OLD
ONES. IF AGREEMENT IS NOT SUFFICIENT, IT REPLACES THE PRESENT
GLOBAL MATRIX WITH THE OLD, UNMODIFIED ONE AND CAUSES
THE ITERATIONS TO BEGIN AGAIN, BUT WITH THE NEW VALUES
OF LOCAL TEMPERATURE DIFFERENCE.

CALL CLOSE(F,TEM,NMF,NENFC,NCANPO,ZA,ZAST,NCCUN,NCHK
1,NUC,NLC)
IF(NCCUN.EC.51) GC TC 103
IF(NCHK.EC.5) GC TC 3
NST=1
CALL FLCCC(XFFG,RFCL,RFCV,XMUL,XMUV,XKLIQ,XKSOL,PHEE,R,TEM,HOK,
INCELC,KCRR,NCC,TBAR,FF,FEEDBAR,VELST,CSUEP)
CALL FCRA(F,NNP)
FORMAT(1,NCC FOR FCRMC12='I5)
CALL FCRMCD(NNP,ZA,KQRR,NCELC,NSDQ,XLC12,
1,XLC23,FCK,NCC,NUC,NLC)
CALL MCBCE(ZA,F,NNP,NCCUN,NEBC,NBC,FNEBC,NLC,NUC)
CALL LECTIE(ZA,NNP,NLC,NUC,320,F,1,1,0,XL,IER)
EC 541 NR=1,NNP,9

4

474

VAT0241C
VAT0242C
VAT0243C
VAT0244C
VAT0245C
VAT0246C
VAT0247C
VAT0248C
VAT0249C
VAT0250C
VAT0251C
VAT0252C
VAT0253C
VAT0254C
VAT0255C
VAT0256C
VAT0257C
VAT0258C
VAT0259C
VAT0260C
VAT0261C
VAT0262C
VAT0263C
VAT0264C
VAT0265C
VAT0266C
VAT0267C
VAT0268C
VAT0269C
VAT0270C
VAT0271C
VAT0272C
VAT0273C
VAT0274C
VAT0275C
VAT0276C
VAT0277C
VAT0278C
VAT0279C
VAT0280C
VAT0281C
VAT0282C
VAT0283C
VAT0284C
VAT0285C
VAT0286C
VAT0287C
VAT0288C

VATC289C
VATC290C
VATC291C
VATC292C
VATC293C
VATC294C
VATC295C
VATC296C
VATC297C
VATC298C
VATC299C
VATC300C
VATC301C
VATC302C
VATC303C
VATC304C
VATC305C
VATC306C
VATC307C
VATC308C
VATC309C
VATC310C
VATC311C
VATC312C
VATC313C
VATC314C
VATC315C
VATC316C
VATC317C
VATC318C
VATC319C
VATC320C
VATC321C
VATC322C
VATC323C
VATC324C
VATC325C
VATC326C
VATC327C
VATC328C
VATC329C
VATC330C
VATC331C
VATC332C
VATC333C
VATC334C
VATC335C
VATC336C

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190 WRITE(6,202)MR,F(MR)
941 WRITE(7,190) F(MR)
945 FORMAT(F10.5)
C CONTINUE
C FCNRMAT(6(F10.5))

HEATE IS A SUBROUTINE WHICH CHECKS TC INSURE THAT
THE TCTAL HEAT TRANSFERREC THROUGH THE TUBE WALL BALANCES.

CALL HEATE(XKSOL,XKLG,F,X,Y,NOUN,TBAR,YBAR,FH,
1KORR,NCC,NCELC,XLO12)

CUTP IS A SUBROUTINE WHICH CUTPUTS THE X AND Y COORDINATES
AND THE LOCAL TEMPERATURE DIFFERENCE, ECTH TC AN OFFLINE PRINTER
AND TC A LISC IN ORDER TC EE PLOTTED USING AVAILABLE PLOTTING
SLERCUTINES.

CALL CUTP(X,Y,NNP,F)
STOP
FORMAT(15,15,15,15,15,15)
FCNRMAT(15,15,15,15,15,15)
END
SUBROUTINE CVERT(X,Y,R,PTEE,NNP)
DIMENSION R(15),PTEE(50)
REAL * 8 X(50),Y(50)
READ (5,100) NANG
FORMAT(15)
DO 110 J=1,NANG,7
READ(5,111)PTEE(JT),PTEE(JT+2),PTEE(JT+3),PTEE(JT+4)
1,PTEE(JT+5),PTEE(JT+6)
FORMAT(7(F8.3,2X))
WRITE(7,111)PTEE(JT),PTEE(JT+1),PTEE(JT+2),PTEE(JT+3),PTEE(JT+4)
1,PTEE(JT+5),PTEE(JT+6)
C CONTINUE
READ(5,100)INRAD
DO 114 JS=1,INRAD
READ(5,112)R(JS)
FORMAT(F10.5)
WRITE(6,112)R(JS)
C CONTINUE

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1Y(KCRR(JE,2))**2))
XL23=DSCT((X(KCRR(JB,2))-X(KCRR(JB,3)))**2)+((Y(KCRR(JE,2))-
1Y(KCRR(JE,3))**2))
1Y(KCRR(JE,1))**2))
1Y(KCRR(JE,1))**2))
IF(JE .LT. 540) GC TC 17
4=1
17 FORMAT((F20.10),
PER=XL12+XL23+XL31
SPER=PER*
SPER=(SPER*(SPER-XL12)*(SPER-XL23)*(SPER-XL31))
ATRI(70)=1,NCO
LC INCELC(JC) .NE. JB) GC TC 170
XLC12(JC)=XL12
XLC23(JC)=XL23
CONTINLE
IF(JE .LT. 540) GO TC 122
CONTINLE
FORMAT('NCO= ',I5)
RETURN
FORMAT(4(I5))
FORMAT(I5,2X,F18.14)
ENCL
LINE FCRMB (ATRI,NEL,NNP,X,Y,ZA,KCRR,NLC,NUC)
LIMENS ICA KCRR(600,3),ZA(320,25)
REAL * 8 ATRI(600),X(320),Y(320),A(3,3)
DOUBLE PRECISION B1,E2,B3,C1,C2,C3,ATR4
NAX=NLC+NLC+1
CC 17C KA=1,NNP
CO 171 KB=1,NAK
ZA(KA,KE)=C.
CONTINLE
C=1,NEL
B1=Y(KCRR(JC,3)))-Y(KCRR(JC,3))
B2=Y(KCRR(JC,1)))-Y(KCRR(JC,1))
E2=Y(KCRR(JC,1)))-Y(KCRR(JC,2))
C2=X(KCRR(JC,3)))-X(KCRR(JC,2))
C3=X(KCRR(JC,2)))-X(KCRR(JC,1))
ATR4=ATRI(C)*4.
KD=JC/8
IF(KC .EG. JC) ATR4=ATR4*200.
WRITE(4,F20.15)
FORMAT((F20.15)
A(1,1)=(B1+B2+C1*C1)/ATR4
A(1,2)=(B1+B2+C1*C2)/ATR4
A(1,3)=(B1+B2+C1*C3)/ATR4
A(2,2)=(B2+B2+C2*C2)/ATR4

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357 DIMENSION PHEE(50),R(15),NCELO(40),DTAL(40)
      1,KCRR(600,3),TBAR(40),F(600),FEEBAR(40)
      1,RCUT=R(1)
      1,FCRMT( F13,2,2(F10.5),2(F20.5),4(F10.5))
      1,PHFHC=C.
      1,CO=310.
      1,NL=1,ACG
      1,TBAR(NL)=(TEM(KORR(NCELC(NL),1))+TEM(KORR(NCELO(NL),2)))/2.
      1,IF(NCAT.NE.0)GC=35
      1,CELC=((X*ML*RCUT*X*KLIC*SCRT(VELST*RCUT*RHCV/XMUV)
      1*TBAR(NL))/(1.7432*FHCV*VELST*VELST*(X+FG+3.54*CSUEF
      1*TEAF(NL))))*.3333
      1,WRITE(6,318) DELC
      1,FCRMT(6,318) DELC
      1,NCAT=N(CAT+1)
      1,FORMAT(1,F14.9)
      1,FEEDBAR(NL)=(PHEE(NL)+FHEE(NL+1))*3.141592654/(180.*2.)
      1,CELFEE=AES(PHEE(NL)-FHEE(NL+1))*3.141592654/180.
      1,TAL(NL)=5*RHCV*VELST*VELST/SCRT(VELST*FHCV/XMUV)
      1*(6.573*FEEBAR(NL)-2.732*FEEBAR(NL))*3+.252*FEEBAR
      1(NL))*5-.0183*FEEBAR(NL))*7+.000043*FEEBAR(NL))*9-.000115*
      1(FEEBAR(NL))*11)
      1,CTAU(NL)=5*RCOV*VELST*VELST/SCRT(VELST*FHCV*RCUT/XMUV)
      1*(6.573-5.156*FEEBAR(NL))*2+1.46*FEEBAR(NL))*4
      1-.1281*FEEBAR(NL))*6+.00387*FEEBAR(NL))*8-.001265*FEEBAR(NL)
      1*10)
      1,IF(FEEBAR(NL).GT.1.884)TAU(NL)=0.
      1,IF(FEEBAR(5),NL,TAL(NL),CTAU(NL)
      1,FORMAT(15,'SHEAR=',F10.5)
      1,IF(FEEBAR(NL).GT.1.884),GC TO 213
      1,CLFLS=((RCUT*X*KLIC*TBAR(NL)*XMUL/(CELC**2*(XHFG+3.54*CSUEP*TBA
      1,FACT1=(5.*X*ML*RCUT*X*KLIC*TBAR(NL)))/((DELC*DTAU(NL))/(2.*TAU(NL)))*CELFEE
      1*(X+FG+3.54*CSUEP*TBAR(NL)))
      1,FACT2=SIN(FEEBAR(NL))*CELO**4
      1,FACT3=3.*TAL(NL)*DELC**2/(5.81*(RHOL-RHCV))
      1,FACT4=3.*TAL(NL)*DELC**3
      1,FACT5=3.*CCS(FEEBAR(NL))*DELO**3
      1,DELCAC=((FACT1-FACT2-FACT3)/(FACT4-FACT5))*CELFEE
      1,WRITE(6,318) DELAD,CLFLS
      1,FCRMT(6,318) DELAD=,F14.5, CLPLUS = ,F14.5)
      1,DELC=DELCAD/2.+DELC
      1,PH(NCELC(NL))=X*KLIC/CELO
      1,PH(NCELO(NL))=X*KLIC/CELO
      1,PHK=PH+CELFEE*F(NCELC(NL))/XKSOL
      1,ANGF(NCELC(NL))=HH(NCELC(NL))*180./3.141592654
      1,WRITE(6,322)NCELO(NL),TBAR(NL),ANGBR,PHK(NCELC(NL)),

```



```

1 F(N(CELC(NL)),DELC
DELO=DELAC/2.+DELC
FORMAT(15,5(F20.9,2X))
CONTINUE
FAVAV=FAH/3.141592654
WRITE(6,343)FAVAV
FORMAT(1,1) AREA WEIGHTED AVERAGE H IS: ',F12.3)
RETURN
FORMAT(15,15)
END
SUBROUTINE FORMOD (NAP,ZA,KCRR,NCELO,NSEC,XLC12,
1 XLC23,FCK,ACC,NUC,NLC)
DIMENSION ZA(320,25),KCRF(600,3),NCELO(40),NSCO(40),HOK(600)
REAL X, XLC12(40),XLC23(40)
FORMAT(1,1) IN FORMOD ',4(I5)
DO 185 JR=1,NCO
FORMAT(2X,15,2(F14.5))
FCK=FCK(NCELC(JR))*XLC12(JR)/3.
XNUM=FCK(JR).EQ.2) GC TC 185
1ZA(KCRR(NCELC(JR),1),(NLC+1))=ZA(KCRR(NCELO(JR),1),(NLC+1))+XNUM
1ZA(KCRR(NCELC(JR),1),(KCRR(NCELO(JR),2)-KCRR(NCELC(JR),1)+NLC+1))=
1ZA(KCRR(NCELC(JR),1),(KCRR(NCELO(JR),2)-KCRR(NCELC(JR),1)+NLC+1))
1+XNUM/2.
1ZA(KCRR(NCELO(JR),2),(NLC+1))=ZA(KCRR(NCELC(JR),2),(NLC+1))+XNUM
1ZA(KCRR(NCELC(JR),2),(KCRR(NCELO(JR),1)-KCRR(NCELC(JR),2)+NLC+1))=
1ZA(KCRR(NCELC(JR),2),(KCRR(NCELO(JR),1)-KCRR(NCELC(JR),2)+NLC+1))
1+XNUM/2.
CONTINUE
RETURN
END
SUBROUTINE MCDBCE (ZA,F,ANP,ACCUN,NEEC,NBC,FNEBC,NLC,NUC)
1 DIMENSION ZA(320,25),F(320),NEBC(40),FNEBC(40)
IF(NCCUN.EC.0) GC TC 801
WRITE(6,788) ACCUN
FORMAT(2X,15)
NAX=ALC(NLC+1)
DO 148 IT=1,NBC
LO 144 JC=1,NAX
1ZA(NEBC(IT),IC)=0.
1ZA(NEBC(IT),(NLC+1))=1.
F(NEBC(IT))=FNEBC(IT)
CONTINUE
FORMAT(F10.5)
RETURN
END
SUBROUTINE CLOSE(F,TEM,NNP,NBNPO,NCANPO,ZA,ZAST,NCCUN,NCFK
1,NLC,NLC)
DIMENSION F(320),TEM(320),NBNPO(40),ZA(320,25),ZAST(320,25)

```



```

322 11=1,NNP
321 321 2(F10.5)
CC FORTINLE
CC 2=4 11=1,NCANFC
233 FORTAT(15)
234 FORTINLE
756 NCFK=C
FORMAT('IN CLOSE' )
CO 705 R=1,NCANFC
JT=NCANFC-JR+1
IF(AES(F(NENFC(JT)))-TEM(NBNPC(JT))) -LT,.C2) GO TC 705
TEM(NENFC(JT))=(TEM(NENFC(JT))+F(NBNPC(JT)))/2.
705 NCFK=1
CC FORTINLE
DOM 820 C=1,NNP
836 TEM(JC)=F(JC)
904 NCCUN=NCCUN+1
NAX=NLC+NLC+1
LC 690 N1=1,NNP
CC 691 N2=1,NAX
ZC(M1,M2)=ZAST(M1,M2)
251 CC FORTINLE(6,15)NCCUN
108 FORTINLE(6,15)
RETURN
END
SUBROUTINE CUTP(X,Y,NP,F,FNEBC)
REAL * 8 X(320),Y(320)
DIMENSION F(320),FNEBC(40)
WRITE(6,158)
DO 163 IL=1,NP
IF (F(IL) .EQ. 45.) GO TO 163
WRITE(6,162)X(IL),Y(IL),F(IL)
WRITE(7,162)X(IL),Y(IL),F(IL)
CC FORTINLE
CC FORTAT(3(F11.5))
163 FEM SCLN')
158 Y-CIST
162 FEM SCLN')
RETURN
END
SUBROUTINE HEATB(XKSCL,XKLIQ,F,X,Y,NCCUN,TBAR,YBAR,HH,
1 KGRF,NCC,NCELC,XLC12)
DIMENSION F(320),TEAR(40),NCELC(40),
1 FH(600),KCR(320,3),NCCBF(40),NCCAP(40)
1,AFEA(40),AFEA(40),QCIN(40),QCUT(40),CCNV(40),CCNA(40)
1,REAL * 8 XLC12(40),X(320),Y(320)
FREAD(5,861) NCCP
FORMAT(15)
DO 860 J=1,NCCP
861

```

```

VAT05777C
VAT05780C
VAT05790C
VAT05800C
VAT05810C
VAT05820C
VAT05830C
VAT05840C
VAT05850C
VAT05860C
VAT05870C
VAT05880C
VAT05890C
VAT05900C
VAT05910C
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VAT05980C
VAT05990C
VAT06000C
VAT06010C
VAT06020C
VAT06030C
VAT06040C
VAT06050C
VAT06060C
VAT06070C
VAT06080C
VAT06090C
VAT06100C
VAT06110C
VAT06120C
VAT06130C
VAT06140C
VAT06150C
VAT06160C
VAT06170C
VAT06180C
VAT06190C
VAT06200C
VAT06210C
VAT06220C
VAT06230C
VAT06240C

```



```

860 READ (5,862) NOBCP(J),NCEAF(J)
862 CCNT=CCNT+1
866 CFORMAT(I5,I5)
      ATCTI=C.
      ATCTI=C.
      CC 864 JA=1,NCO
      CF=(F(NCBOF(JA))+F(NCBOF(JA+1)))/2.-(F(NCBAF(JA))+F(NCBAF(JA
1+1)))/2.
1 DIS1=DSQRT((X(NOBCF(JA))-X(NCBAF(JA)))*2+(Y(NOBCP(JA))-Y(NOBA
1(JA)))*2)
1 FFORMAT(3X,2(F10.7))
  IF(JA .GT. 1) GO TC 870
  AREA(JA)=.5*(DSQRT((X(NCBOF(JA))-X(NOBCP(JA+1)))*2
1+(Y(NCBOF(JA))-Y(NCBOF(JA+1)))*2))
  GO TC 872
  IF(JA .EQ. NBOF) GO TC 871
  AREA(JA)=.5*(DSQRT((X(NCBOF(JA))-X(NOBCP(JA+1)))*2
1+(Y(NCBOF(JA))-Y(NCBOF(JA+1)))*2))
  1+(Y(NCBOF(JA))-Y(NCBOF(JA+1)))*2)
  1+(Y(NCBOF(JA-1))-X(NCBOF(JA)))*2
  1+(Y(NCBOF(JA-1))-Y(NCBOF(JA)))*2)
  CC TC 872
  AREA(JA)=.5*(DSQRT((X(NOBCF(JA-1))-X(NCBOF(JA)))*2
1+(Y(NCBOF(JA-1))-Y(NCBOF(JA)))*2))
  CINT(JA)=XKSC*AREA(JA)*CF/CIST
  CICTI=CICTI+CIN(JA)
  ATCTI=ATCTI+AREA(JA)
  CCNT=CCNT+1
  CFORMAT(I5,I5)
  DC 870 JA=1,NCCP
  READ (5,872) NOCOP(J),NCCAP(J)
  CCNT=CCNT+1
  CFORMAT(I5,I5)
  ATCTI=C.
  ATCTI=C.
  CC 874 JA=1,NCO
  CF=(F(NCBOF(JA))+F(NCCAF(JA+1)))/2.-(F(NCCCF(JA))+F(NCOCOP(JA
1+1)))/2.
1 DIS1=DSQRT((X(NOCOP(JA))-X(NCCAF(JA)))*2+(Y(NCOCOP(JA))-Y(NCOCAP(JA
1)))*2)
1 FFORMAT(3X,2(F10.7))
  IF(JA .GT. 1) GO TC 875
  AREA(JA)=.5*(DSQRT((X(NCOCOP(JA))-X(NCCCF(JA+1)))*2
1+(Y(NCOCOP(JA))-Y(NCCCF(JA+1)))*2))
  GO TC 876
  IF(JA .EQ. NCCP) GO TC 851
  AREA(JA)=.5*(DSQRT((X(NCCCF(JA))-X(NCOCOP(JA+1)))*2

```



```

1+(Y(NCCCCF(JA))-Y(NCCCCF(JA+1)))*2))
1+.5*(CSCRT((X(NCCCCF(JA-1)))-X(NCCCCF(JA)))*2)
1+(Y(NCCCCF(JA-1))-Y(NCCCCF(JA)))*2))
CC TC 552
1+AREA C(JA)=.5*(DSCRT((X(NCCCCF(JA-1)))-X(NCCCCF(JA)))*2)
CCUT(JA)=XKSCCL*AREAC(JA)*CF/DIST
CTCTC=CTCTC+CCUT(JA)
ATCTC=ATCTC+AREAO(JA)
ATTCT=AREAC(JA)*TBAR(JA)+ATTCT
CONTINLE
ATATC=C.
CTCTC=C.
CC 88C JB=1, NCO
CCCNV(JE)=FF(NCELO(JE))*XL012(JB)*TBAR(JB)
CCNA(JE)=FF(NCELO(JE))*TEAR(JB)
ATCTC=ATCTC+XL012(JB)
CTCTC=CTCTC+CCGNV(JE)
CONTINLE
WRITE(6,875) AREA CUT C CCND OUT LOCAL FLX ' '
FCFMA CCNV Q CCNCESEC AREA IN Q CCND IN
1+AREA CCNV Q CCNCESEC LOCAL FLX ' '
CC 878 JC=1, NCO
WRITE(6,877) AREAI(JC), CIN(JD), AREAO(JC), CCUT(JD), XL012(JC), QCONV
1(JC), CCNA(JC)
CONTINLE
FORMAT(6,F12.6,1X),F13.2)
WRITE(6,882)
FCFMA ' '
WRITE(6,875)
WRITE(6,877) ATOTI, CTCTI, ATOTO, QTOTO, ATGTC, CTCTC
TAVG=ATCTI/ATOTO
WRITE(6,40C)TAVG
WRITE(6,40C)TAVG
FORMAT( ' AVERAGE WALL TENF = ',F10.5)
FETURN
END
551
952
544
88C
875
878
877
882
40C

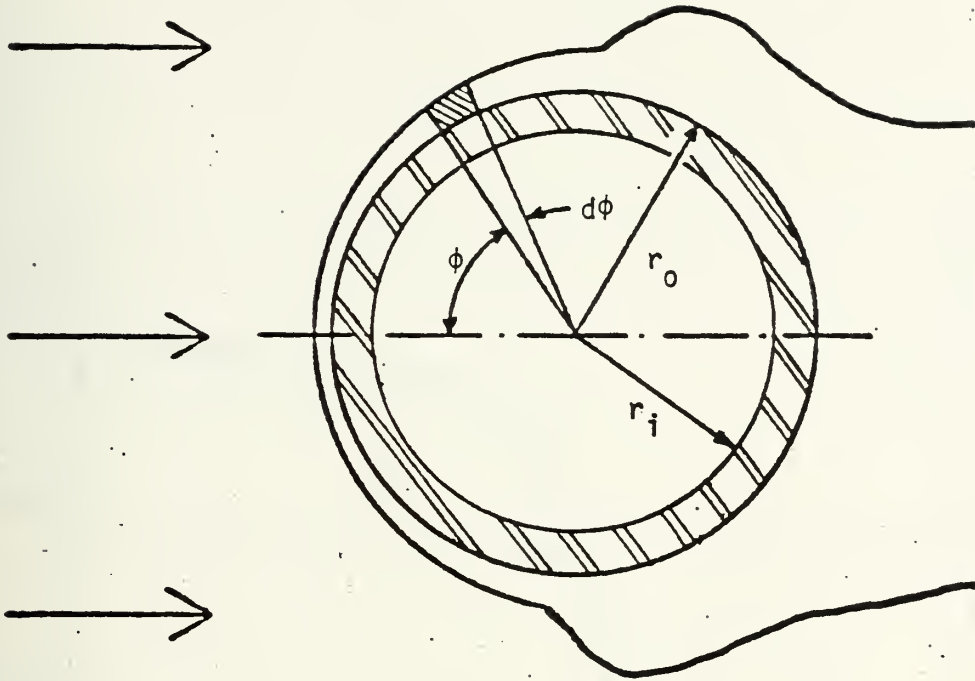
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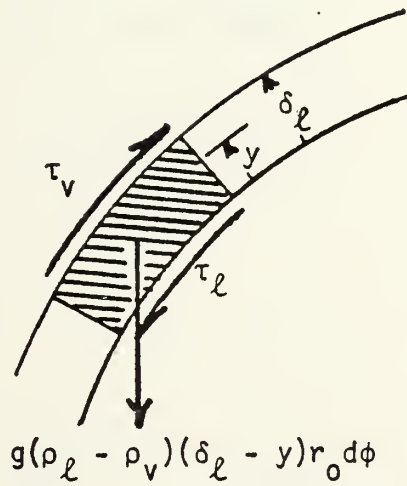
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VAT0690C
VAT0691C
VAT0692C
VAT0693C
VAT0694C
VAT0695C
VAT0696C
VAT0697C
VAT0698C
VAT0699C
VAT0700C
VAT0701C
VAT0702C
VAT0703C
VAT0704C
VAT0705C
VAT0706C
VAT0707C
VAT0708C
VAT0709C

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APPENDIX C



(a)



(b)

Figure 1. Schematic diagrams of physical model.

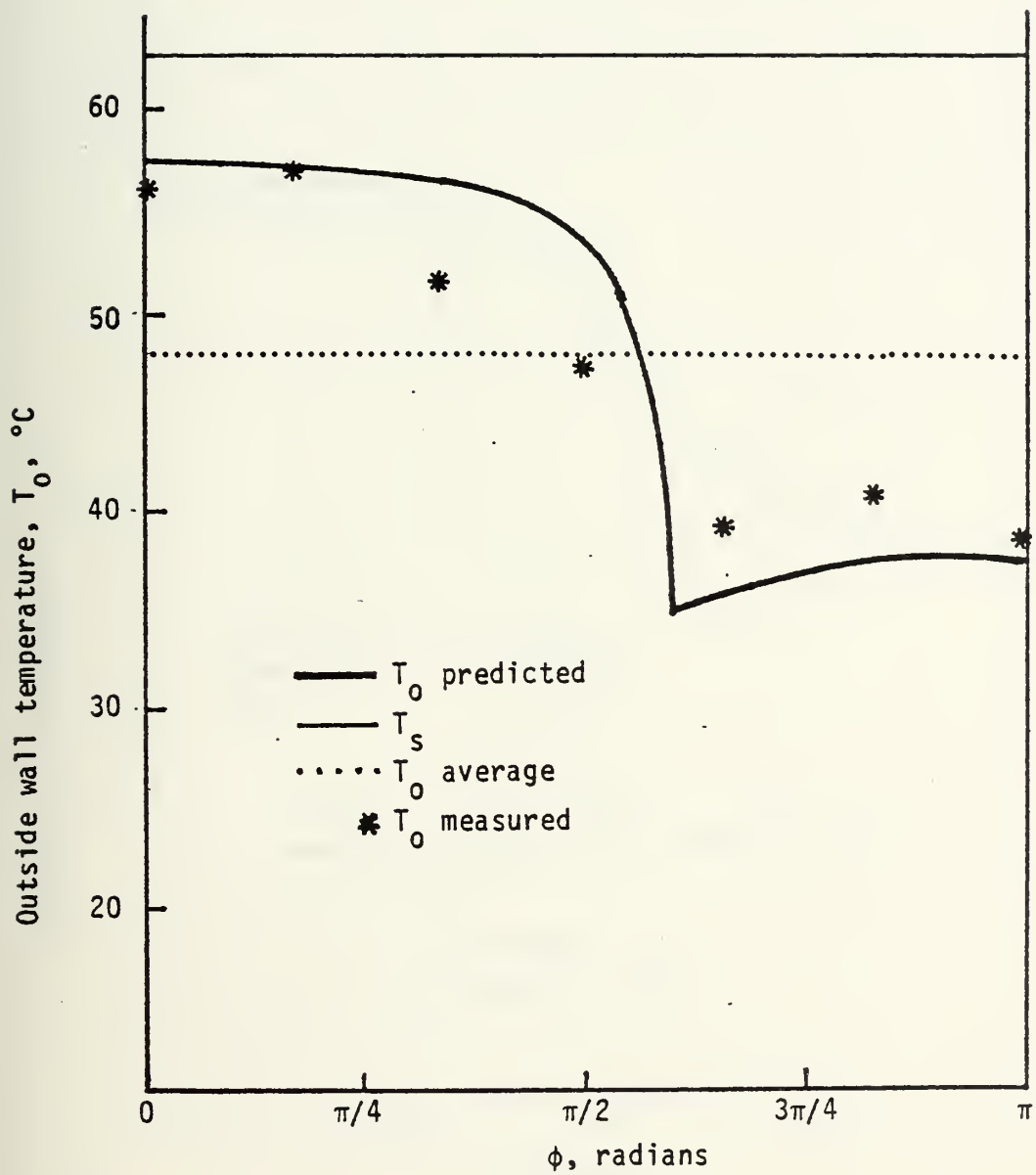


Figure 2. Predicted and measured T_o versus angle from stagnation point [18].

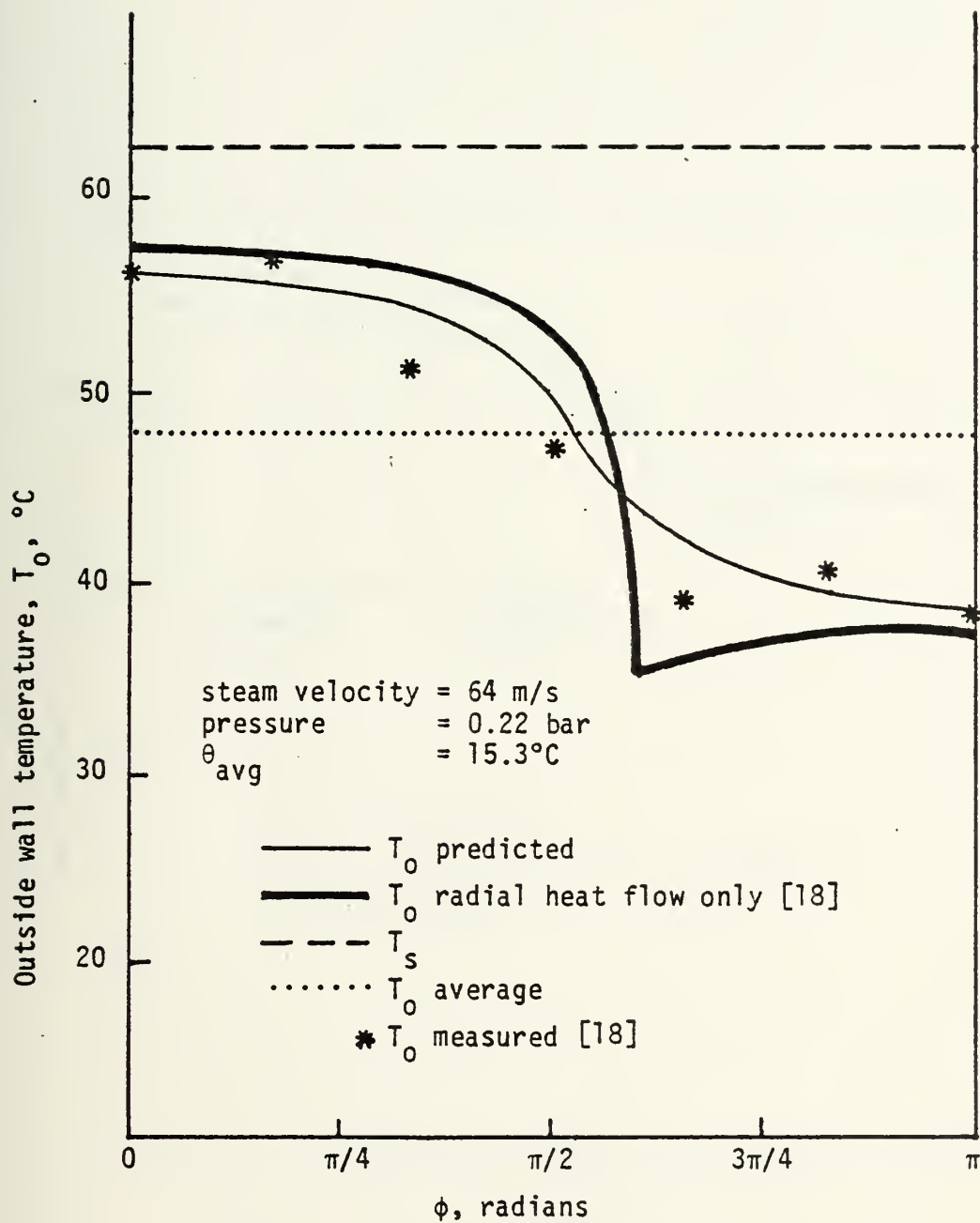


Figure 3. Predicted and measured T_o versus angle from forward stagnation point.

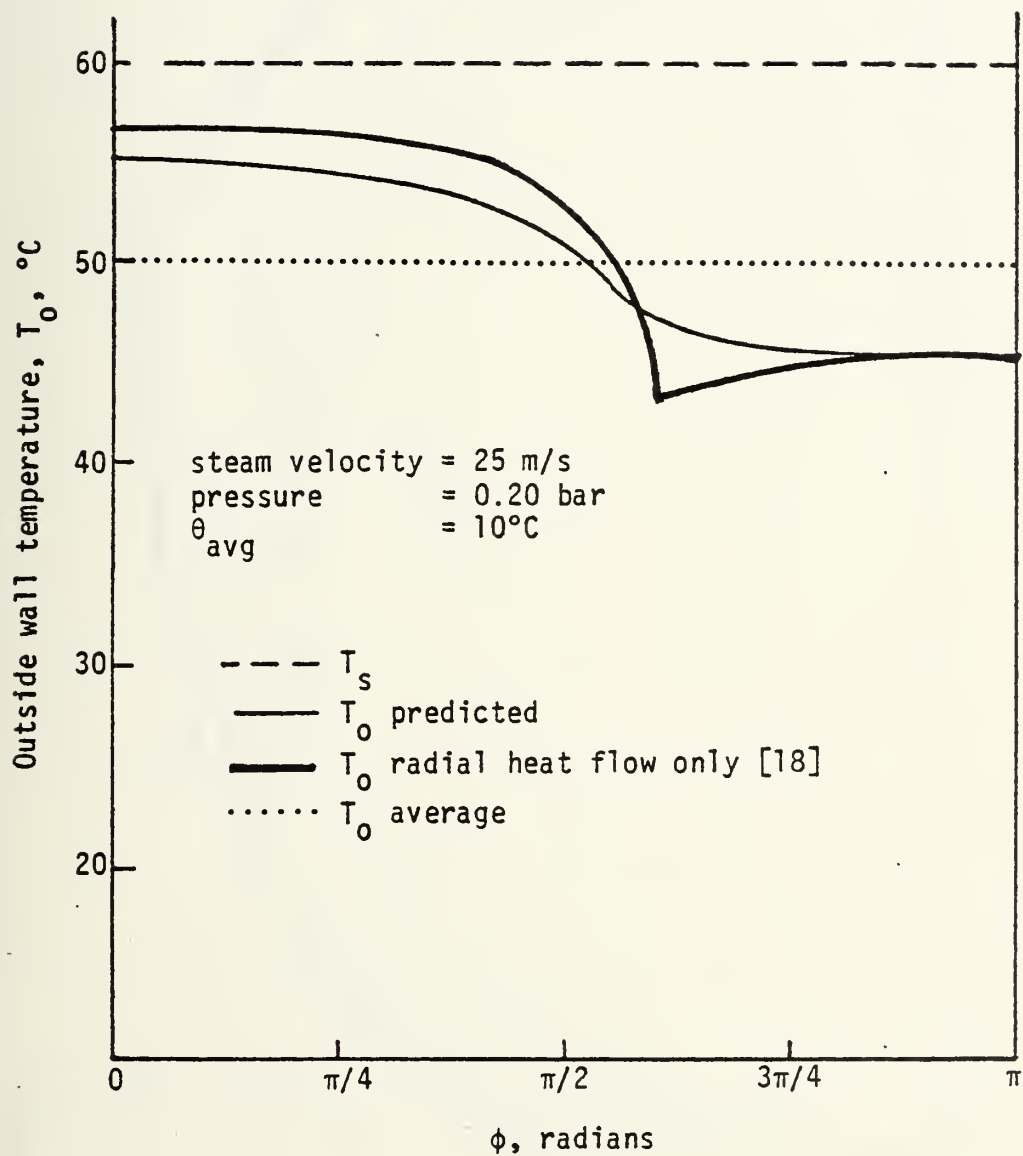


Figure 4. Predicted T_o versus angle from forward stagnation point.

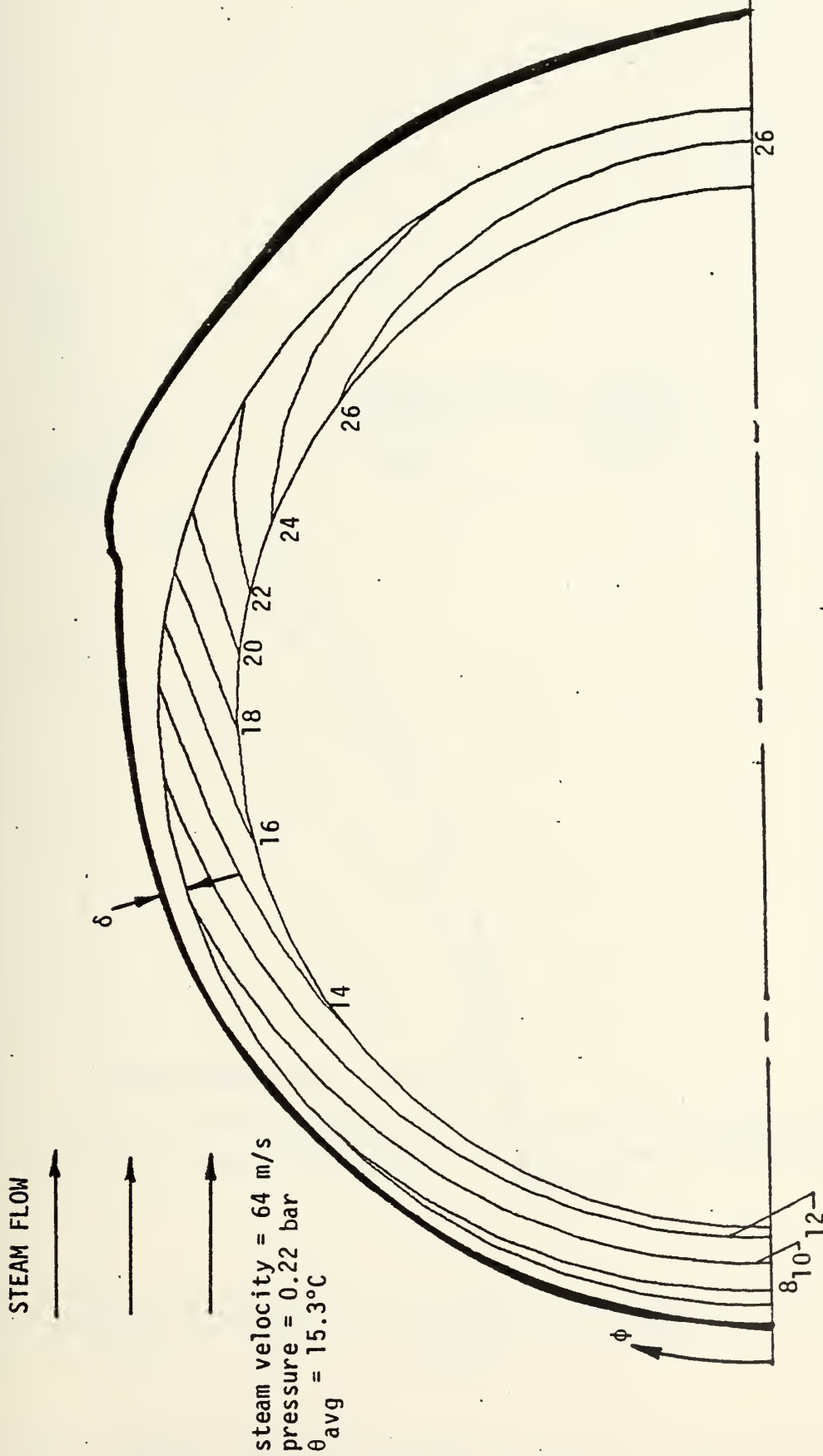


Figure 5. Isotherms (in degrees Celsius) of θ inside tube wall for top half of tube. Thickness of condensate film is shown with a magnification of 25 times.

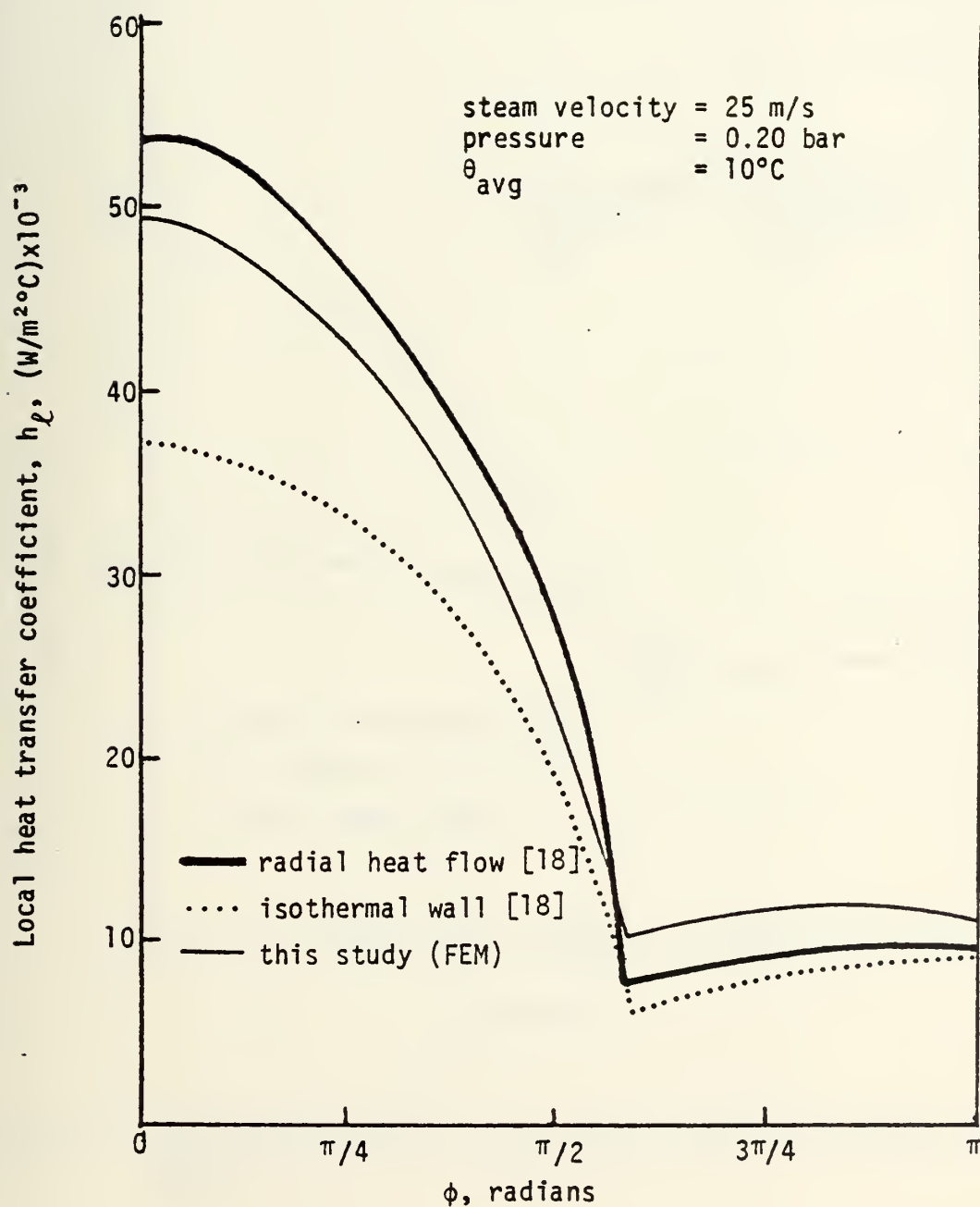


Figure 6. Predicted local heat transfer coefficient versus angle from forward stagnation point.

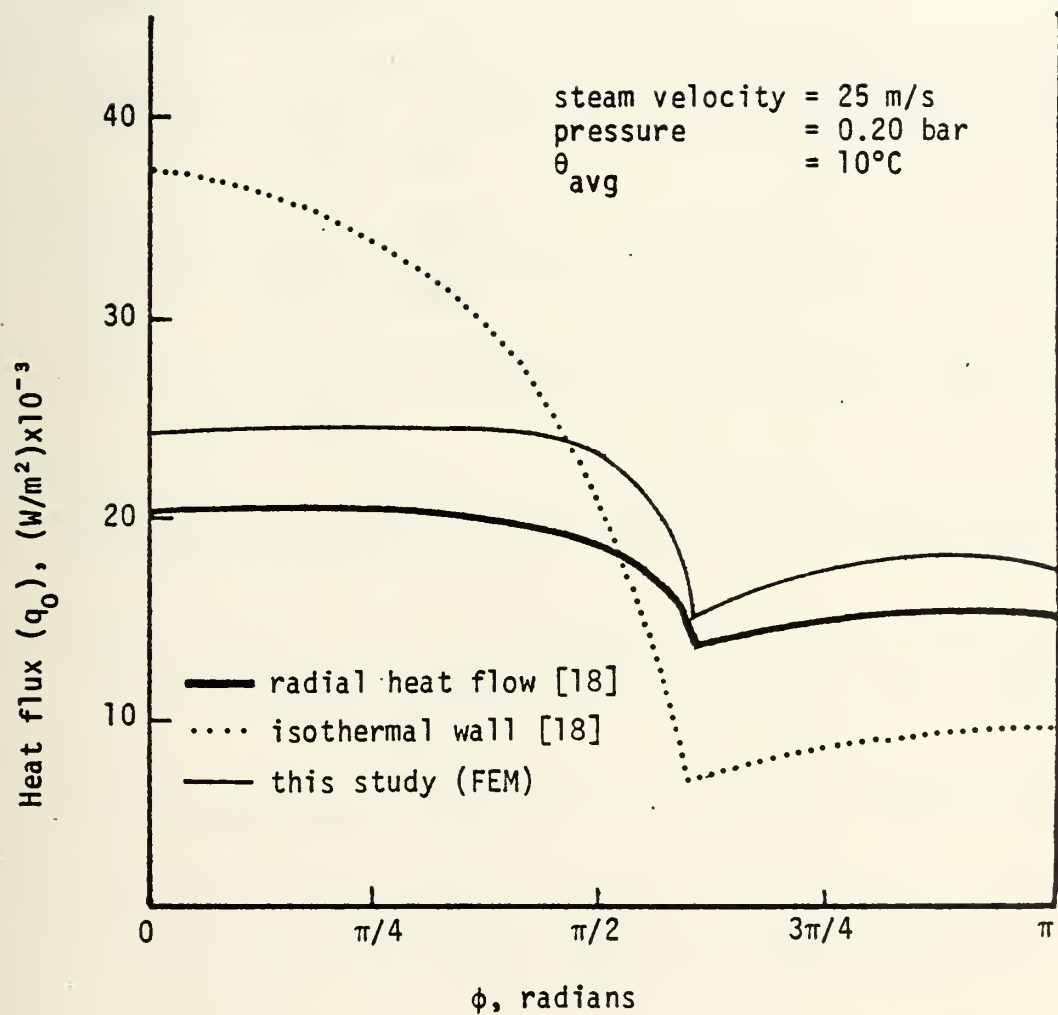


Figure 7. Predicted local heat flux versus angle from forward stagnation point.

LIST OF REFERENCES

1. Standards for Steam Surface Condensers, 6th ed., Heat Exchange Institute, 1970.
2. Standards of Tubular Exchanger Manufacturers Association, 4th ed., Tubular Exchanger Manufacturers Association, Inc., 1959.
3. Department of the Navy, Design Data Sheet Section DOS4601-1, Steam Condensers, 15 October 1953.
4. Collier, J. G. and Wallis, G. B., Two Phase Flow and Heat Transfer, lecture notes presented for a summer course, July 24 - August 4, 1967, Stanford, California, Volume 4, pp 1210 - 1215, 1967.
5. Rohsenow, W. M., Webber, J. H., and Ling, A. T., "Effect of Vapor Velocity on Laminar and Turbulent - Film Condensation," Trans. ASME, V. 78. pp 1637 - 43, 1956.
6. Sparrow, E. M. and Gregg, J. L., "A Boundary Layer Treatment of Laminar Film Condensation," Trans. ASME, V. 81C, pp 13 - 18, 1959.
7. Koh, J. C. Y., Sparrow, E. M., and Hartnett, J. P., "The Two-Phase Boundary Layer in Laminar Film Condensation," Int. J. Heat Mass Transfer, V. 2, pp 69 - 82, 1961.
8. Shekriladze, I. G. and Gomelaury, V. I., "Theoretical Study of Laminar Film Condensation of Flowing Vapor," Int. J. of Heat Mass Transfer, V. 9, pp 581 - 591, 1966.
9. Denny, V. E. and Mills, A. F., "Non-Similar Solutions for Laminar Film Condensation on a Vertical Surface," Int. J. of Heat Mass Transfer, V. 12, pp 965 - 979, 1969.
10. Mayhew, Y. R., Griffiths, D. J., and Phillips, J. W., "Effect of Vapor Drag on Laminar Film Condensation on a Vertical Surface," Proc. Instn. of Mech. Engrs., V. 180 (3J), pp 280 - 289, 1965 - 66.
11. Mayhew, Y. R. and Aggarwal, J. K., "Laminar Film Condensation with Vapor Drag on a Flat Surface," Int. J. of Heat Mass Transfer, V. 16, 1944 - 49, 1973.

12. Denny, V. E. and South III, V., "Effects of Forced Flow, Noncondensables, and Variable Properties on Film Condensation of Pure and Binary Vapors at the Forward Stagnation Point of a Horizontal Cylinder," Int. J. of Heat Mass Transfer, V. 15, pp 2133 - 2142, 1972.
13. Nicol, A. A. and Wallace, D. J., "Condensation With Appreciable Vapor Velocity and Variable Wall Temperature," Steam Turbine Condensers, NEL Report No. 619, pp 27 - 38, 1976.
14. Silver, R. S., "An Approach to a General Theory of Surface Condensers," Proc. Instn. Mech. Engrs., V. 178, pp 339 - 376, 1963 - 64.
15. Wallis, G. B., "Use of the Reynolds Flux Concept for Analyzing One-Dimensional Two Phase Flow," Int. J. of Heat Mass Transfer, V. 11, pp 445 - 458 and 459 - 472, 1968.
16. Fujii, T., Vehara, H. and Kurata, C., "Laminar Filmwise Condensation of Flowing Vapor on a Horizontal Cylinder," Int. J. Heat Mass Transfer, V. 15, 235 - 246, 1972.
17. Nobbs, D. W. and Mayhew, Y. R., "Effect of Downward Vapor Velocity and Inundation on Condensation Rates on Horizontal Tube Banks," Steam Turbine Condensers, NEL Report No. 619, pp 39 - 52, 1976.
18. Nichol, A. A., Bryce, A., and Ahmed, A. S. A., Condensation of a Horizontally Flowing Vapor on a Horizontal Cylinder Normal to the Vapor Stream, paper presented at Sixth International Heat Transfer Conference, Toronto, Canada, August 7 - 11, 1978.
19. Fujii, T., Honda, H. and Oda, K., "Condensation of Steam on a Horizontal Tube - The Influence of Oncoming Velocity and Thermal Conduction at the Tube Wall,".
20. Fujii, T., Uehara, H., and Oda, K., "Filmwise Condensation on a Surface with Uniform Heat Flux and Body Force Convection," Heat Transfer - Japanese Research, V. 1, no. 4, pp 76 - 83, 1972.
21. Holman, J. P., Heat Transfer, p 204, McGraw - Hill, 1976.
22. Schlichting, H., Boundary Layer Theory, p 160, McGraw - Hill, 1968.
23. Ozisik, M. N., Boundary Value Problems of Heat Conduction, pp 338 - 345, International Textbook Co., 1968.

24. Fairweather, G., Finite Element Galerkin Methods for Differential Equations, pp 21 - 66, Marcel Dekker, Inc, 1978.
25. Carslaw, H. S. and Jaeger, J. C., Conduction of Heat in Solids, pp 166 - 168, Oxford: Clarendon Press, 1959.
26. Mills, A. F., Tan, C., and Chung, D. K., "Experimental Study of Condensation from Steam-Air Mixtures Flowing Over a Horizontal Tube: Overall Condensation Rates," Proc. Fifth Intl. Heat Transfer Conf., pp 20 - 23, 1974.

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